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# Effect of boundry layer control, through surface holes, on the film coefficient of convective heat transfer

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EFFECT OF BOUNDARY LAYER CONTROL, THROUGH SURFACE HOLES, ON THE FILM COEFFICIENT OF CONVECTIVE HEAT TRANSFER

BY

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THESIS

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# I. INTRODUCTION

This thesis has as its objective, the effect of some boundary layer control on the convective heat transfer film coefficient, metal to liquid. The boundary layer control was produced by drilling 1/16 inch diameter holes on the tube surface.

In this modern world of atoms and the current development of large quantities of its energy at low cost, the subject of designing heat exchange devices of higher efficiency is receiving considerable attention. In a reactor, or any type of heat exchange device, there are limits for the temperatures of a heat exchange surface because of corrosion, strength of metals etc. The high heat release rates of $5X10^5$ Btu/hour sq. foot that are common in nuclear reactors have made obsolete the design practices of the boiler industry which considers a heat flux of $1X10^5$ Btu/hour sq. ft. as a maximum rate.

Forced convection is extensively used to-day in existing reactors as a means to transfer heat energy generated within the reactor. Some fluid flows through the reactor in passages arranged to provide heat extraction distributed throughout the reactor. In order to get the maximum power from the system, it is essential to design the heat extraction system to remove maximum possible energy with a minimum expenditure of energy for pumping the fluid. If less fluid is pumped for the same extraction of energy, the temperature (outlet) of the fluid will be high and so the energy available will be at a higher potential.

These objectives can be achieved with a higher efficiency of heat transfer. The higher efficiency of heat transfer means a device to increase the film coefficient of convective heat transfer which is entirely dependent upon the size and nature of the surface film.

This particular investigation was chosen by the author because the control of boundary layer by suction through the slots and porous materials is widely used in aircraft industry. It has been noted by several investigators, when mass crosses a boundary layer in a direction perpendicular to the motion of the fluid, the magnitude and direction of the mass transfer effects the properties of the boundary layer. The boundary layer thickness, stability, velocity, temperature, and concentration profiles are altered. At the same time the heat transfer coefficient is changed. In general, mass transfer from the fluid to the wall ("Suction") increases the magnitude of the heat transfer coefficient, while mass transfer from the wall to the fluid ("Blowing") decreases the magnitude of the heat transfer coefficient. The exploitation of these effects has also important applications, to the cooling of gas turbine blades, the development of high lift airfoils and in the industrially important techniques of drying, absorption, extraction, distillation, and adsorption.

The author wishes to express his sincere thanks to Dr. A.J.Miles chairman, Department of Mechanical Engineering, for suggesting this problem, and his valuable guidance in preparing this volume. The author is indebted to Prof. G.L.Scofield, Mechanical Engineering Department, who gave some of his valuable time to guide the author in setting up the experimental apparatus and in writing this thesis. He would also like to express his gratitude to Mr. R.D.Smith, laboratory technician, Mechanical Engineering Department, for the installation of the test facility.

## II. TABLE OF UNITS AND SYMBOLS

| Symbol | Significance and units. |
|---|---|
| $a$ | Cross-sectional area of flow, sq. ft. |
| $A$ | Heat transfer surface area, sq. ft. |
| $C_p$ | Specific heat of water, Btu/lb. °F. |
| $D$ | Equivalent diameter for heat transfer, ft. |
| $G$ | Mass flow rate of water, lbs./hr.ft$^2$. |
| $h$ | Convective heat transfer film coefficient, Btu/hr.ft$^2$. °F. |
| $K$ | Thermal conductivity, Btu/hr.ft$^2$. °F./ft. |
| $m, n, \alpha$ | Constants. |
| $N_u$ | Nusselt's number, $hD/K$, dimensionless. |
| $P_r$ | Prandtl number, $C_p\mu/K$, dimensionless. |
| $q$ | Heat flow, Btu/hr. |
| $R_e$ | Reynold's number, $GD/\mu$, dimensionless. |
| $t$ | Average temperature of tube surface, °F. |
| $T_1$ | Inlet water temperature, °F. |
| $T_2$ | Outlet water temperature, °F. |
| $T$ | Average temperature of water, °F. |
| $\Delta t$ | Difference between the average surface temperature and the average water temperature, °F. |
| $W$ | Mass flow of water, lbs./hr. |
| $\mu$ | Dynamic viscosity of water, lb/hr.ft. |

# III. REVIEW OF LITERATURE

## 1. Discussion of Convective Heat Transfer & Film Theory:-

Heat transfer by convection is a transportation of heat energy by fluid particles which are in motion and when the motion of the fluid is accomplished by mechanical means, it is called forced convection. Since in this process heat is conveyed mechanically from one particle to another, it is obvious that the transfer of energy depends upon the motion of the fluid and is governed by the laws of fluid dynamics, in addition to the laws of heat conduction and heat storage which must be considered at the same time.

The analytical expression for the transfer of heat by convection involves a quantity called "film coefficient" of heat transfer, which is generally written as[1]

$$q = hA\Delta t \qquad \text{---------------(1)}$$

which is sometimes referred to as Newton's cooling law. The proportionality factor, h, is chiefly dependent upon the mechanism of flow, the fluid properties, geometry, and velocity.

The concept has developed that when a fluid flows over a surface, (whether in stream line or in turbulent flow), a stagnant film[2] adheres to the surface and acts as a heat insulator. In fact, where the main fluid stream is moving at a velocity in the range of turbulent flow, the film itself is divided into two layers:

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1. All references are in bibliography.

the first composed of particles completely without motion adhering to the surface and particles creeping along in streamline flow with increasing velocity as the distance from the surface is increased; and the second layer, which is sometimes called a buffer layer, much thicker than the first, being a transition zone composed of eddy currents moving at a higher velocity although not so swiftly as the main portion of the fluid stream. The boundary between these two layers is not sharply defined.

Ordinarily no attempt is made to measure the thickness of the film which may be immeasureably thin or several hundredths of an inch in thickness, but in the study of heat transfer it is visualized as a barrier to the flow of heat, a barrier that adheres to the surface but is partially wiped off and accordingly reduced in effectiveness as the velocity of fluid is increased.

In the process of heat transfer by convection, heat is transferred by conduction through the stagnant portion of the film and is then transferred to the moving particles and carried away by convection currents into the main portion of the fluid stream.

In the study of heat transfer by convection, however, the concept of the film as being completely stagnant appears to be the best approach to the problem. Heat is presumed to be conducted through this film in proportion to the size and shape of the surface, the specific heat and conductivity of the fluid, the difference in temperatures on the two sides of the film, and its thickness which, although not measured, has been found to be dependent on its viscosity and density and on the velocity of the fluid stream.

Extremely delicate studies of the velocity distribution in the film and beyond have since established that the film is not exactly stagnant, but that it is in true viscous motion with a velocity gradient across it, starting with zero velocity at the solid surface. At the same time temperature distributions have shown a straight-line temperature gradient across the film, proving that heat is conducted across the laminar film just as though the film were motionless.

The film coefficient during forced convection has been correlated for any fluid with a $P_r > 0.7$ in turbulent motion by Colburn[3], and expressed by dimensionless numbers in the equation:

$$\frac{hD}{K} = 0.023\left[\frac{GD}{\mu}\right]^{0.8}\left[\frac{\mu c_p}{K}\right]^{n} \qquad \text{----------------(2)}$$

$$\text{or} \quad Nu = 0.023\,[Re]^{0.8}\,[P_r]^{n}$$

n = 0.4 for fluid heating.

n = 0.3 for fluid cooling.

Equation (2) can be written in a more general form as

$$\frac{hD}{K} = \alpha\left[\frac{WD}{a\mu}\right]^{m}\left[\frac{\mu c_p}{K}\right]^{n} \qquad \text{----------------(3)}$$

At the constant average bulk temperature, the values of D, K, a, $\mu$, $C_p$ will be constant, so the equation (3) can be written

$$h = C\,(W)^{m}$$

$$\text{where} \quad C = \alpha\frac{K}{D}\left(\frac{D}{a\mu}\right)^{m}\left(\frac{\mu c_p}{K}\right)^{n}$$

Taking the logarithms of both sides,

$$\log h = \log C + m \log W$$

which reduces on logarithmic coordinates to an equation of the form

$$h = C + mW \qquad \text{----------------(4)}$$

which is necessarily a straight line relation between h and W.

2. Boundary Layer Control by Suction:-

The idea was originally given by Dr. A. A. Grifth of Great Britain, that the boundary layer may be removed by suction through the slots and porous materials, hence the various characteristics of the boundary layer can be controlled. This idea was first used in designing low drag airfoil, where the low drag was obtained by keeping as much of the boundary layer laminar as possible without boundary layer separation before the trailing edge. No literature is available on the effect of this phenomenon on the convective heat transfer particularly for incompressible fluids.

Suction always has a stabilizing effect on the laminar layer.[5] The stabilization effect of the suction results from:

(1) A reduction of the boundary layer thickness.

(2) Change in the shape of the laminar velocity distribution.

(3) Decreasing the value of the boundary layer Reynold's number.

(4) By elimination of the greatly retarded portion of the layer in the neighborhood of the wall with rising pressure, the laminar separation and hence the transition could be avoided even for considerable pressure increase.

The following effects of boundary layer control[6] have been developed:

(1) Reduction of profile drag by eliminating turbulent separation and by increasing the relative extent of laminar flow.

(2) Increase of the maximum lift coefficient through control of laminar and turbulent separation.

(3) The use of suction and blowing slots near the trailing edge of an airfoil as a means of lateral control.

(4) The use of boundary layer control as a means of increasing the efficiency of diffuser and bends.

(5) To influence shock boundary layer interaction at high speed and, in particular, to eliminate boundary layer separation following the shock.

3. Effect of Rectangular Slots on Heat Transfer.

The effect of boundary layer control by cutting 1/16 inch wide rectangular slots on the pipe surface along the circumference, on convective heat transfer coefficient has been investigated by George.[8] The length of each slot was equal to half the total circumference. He has shown that the convective heat transfer coefficient increases from about 100% to 200% compared with no slots on the pipe surface and water flowing inside and outside of the pipe simultaneously, depending on the arrangement and spacing of the slots. He noted that the maximum increase of heat transfer can be obtained by arranging the slots in a longitudinal spiral, with a suitable spacing between them. He also noted that at lower rates of water flow, the effect of this boundary layer control by slots is greater and gives higher percentage increase of convective heat transfer coefficient.

# IV. DISCUSSION

## 1. Apparatus Used in Experiment:-

Figure 1 is a schematic diagram of the apparatus, which shows that the flow of water was taken from a water pipe line which is directly connected with the boiler feed pump. A small water pump was also placed in line which may be used to produce greater flow head if required.

(2) A differential pressure type flowmeter of Minneapolis-Honeywell Regulator Company to measure the flow of water, which is capable of measuring flow up to 4 gallons per minute.

(3) A pressure gage to measure the pressure of inlet water.

(4) A thermocouple installed to measure the temperature of inlet water.

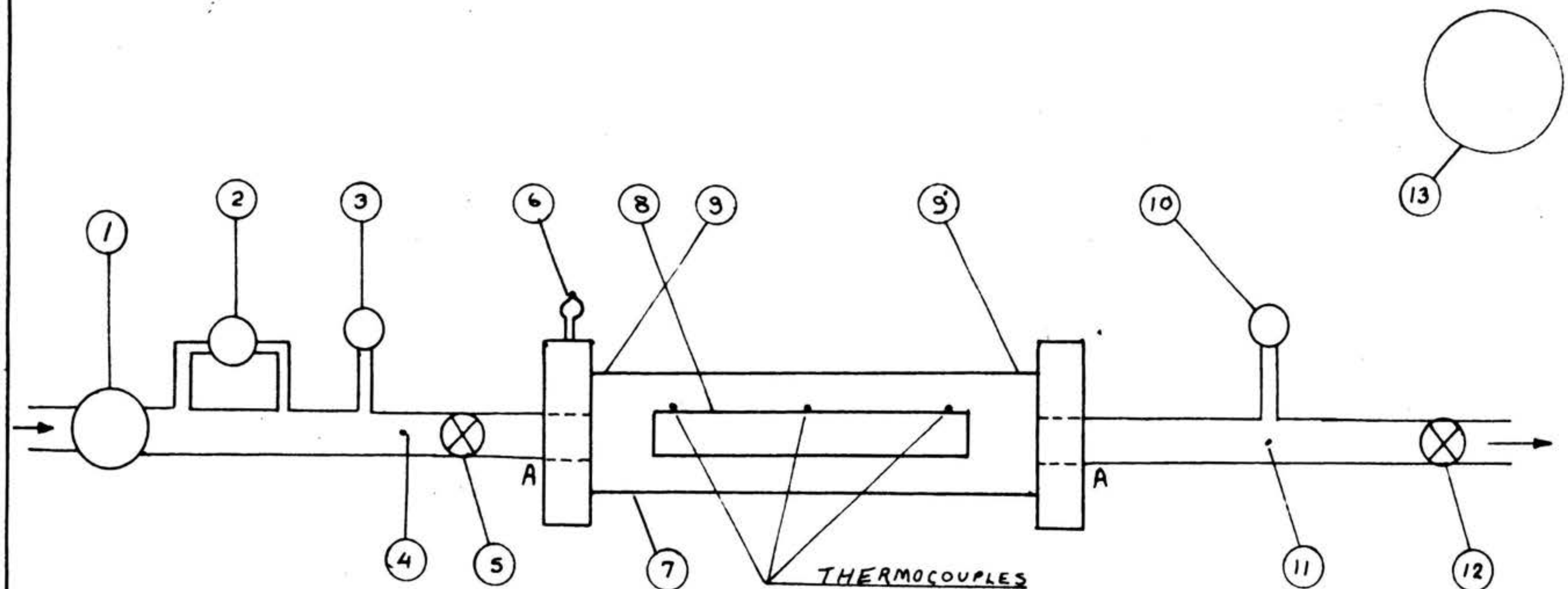
(5) A flow control valve to control the flow.

(6) An air breathing valve to take the air out, which was trapped in the glass tube at the time of starting flow.

(7) A glass tube of 2 inch diameter was placed between two flanges, fastened together by four stay rods. The detailed dimensioned drawing of this fixing has been shown in fig. 2. On this glass tube, an induction heating coil 24 inches long was wrapped to heat the test specimen.

(8) The test specimen was placed inside the glass tube, (the detailed drawings of different test specimens have been

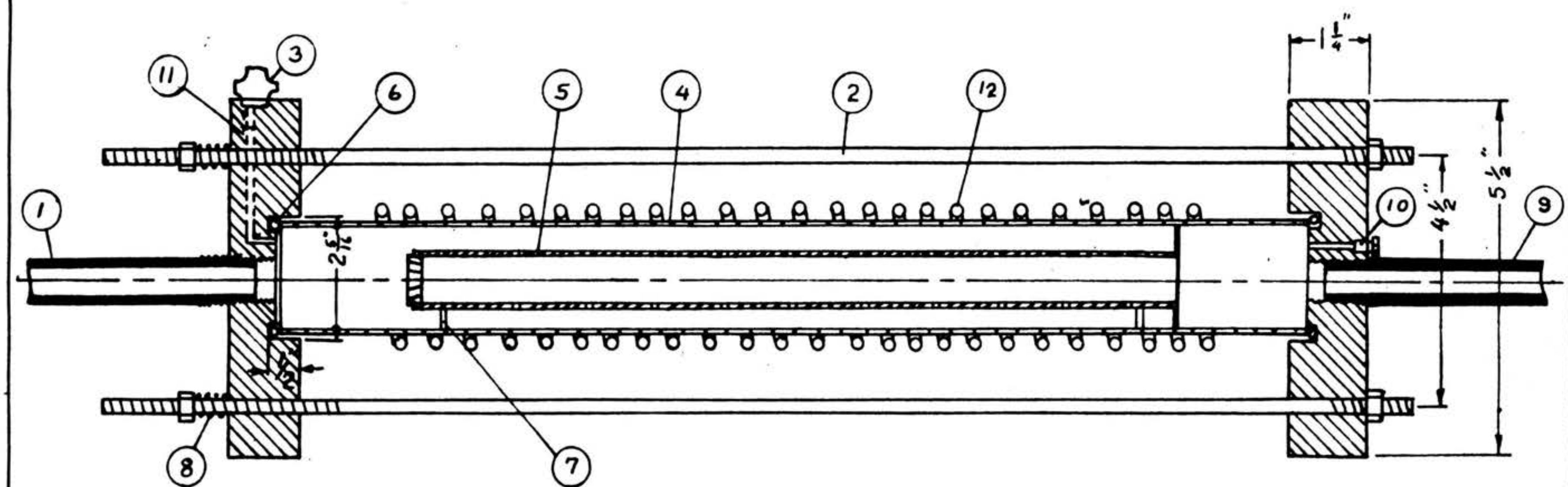
FIGURE # 1




(1) PUMP
(2) FLOWMETER
(3) PRESSURE GAGE
(4) THERMOCOUPLE (INLET WATER)
(5) CONTROL VALVE
(6) AIR BREATHING VALVE
(7) GLASS TUBE
(8) TEST SECTION
(9)(9') HEATING COIL
(10) PRESSURE GAGE
(11) THERMOCOUPLE (OUTLET WATER)
(12) CONTROL VALVE
(13) POTENTIOMETER TO RECORD TEMPERATURES
(A)(A) DETAIL DRAWING IN FIGURE # 2.

GENERAL FLOW DIAGRAM OF APPARATUS

FIGURE # 2

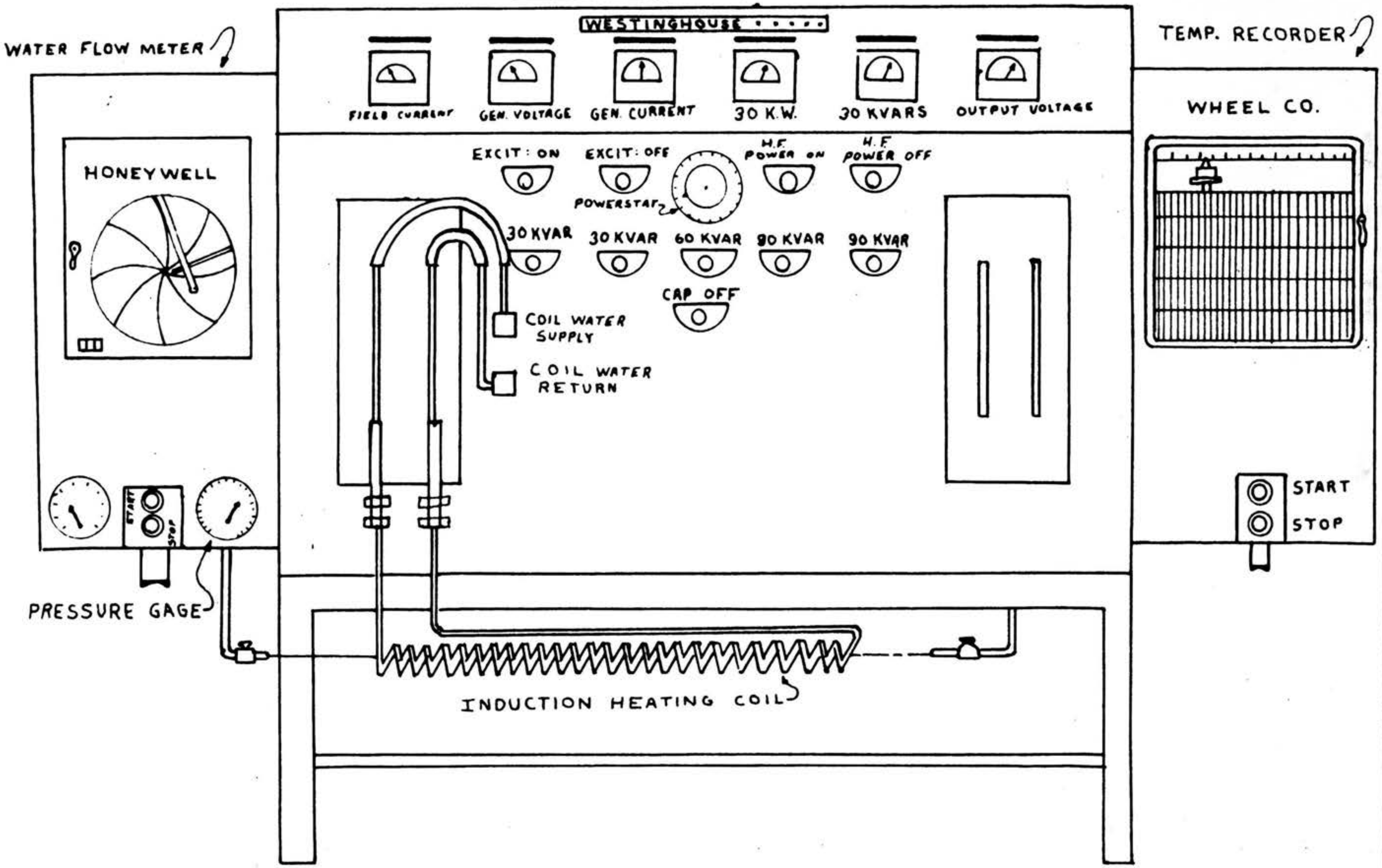



1. 1/2" WATER INLET PIPE
2. STAY RODS (4 EACH) 3/8" x 36"
3. DRAIN COCK FOR AIR
4. GLASS TUBE 30" LONG O.D. 2 1/4" x I.D. 1 7/8"
5. TEST PIECE 20" LONG O.D. 1.05" x I.D. .825"
6. O-RING O.D. 2 3/16" x I.D. 1 7/16"
7. SUPPORT FOR TEST PIECE
8. SPRINGS ON STAY RODS
9. 1/2" WATER OUTLET PIPE
10. PLUG FOR THERMOCOUPLE WIRES
11. ALUMINUM FLANGE
12. HEATING COILS 3/8" DIA. x 25 TURNS

NOTE: DIMENSIONS AS SHOWN, NOT TO SCALE.

SECTIONAL ELEVATION OF APPARATUS

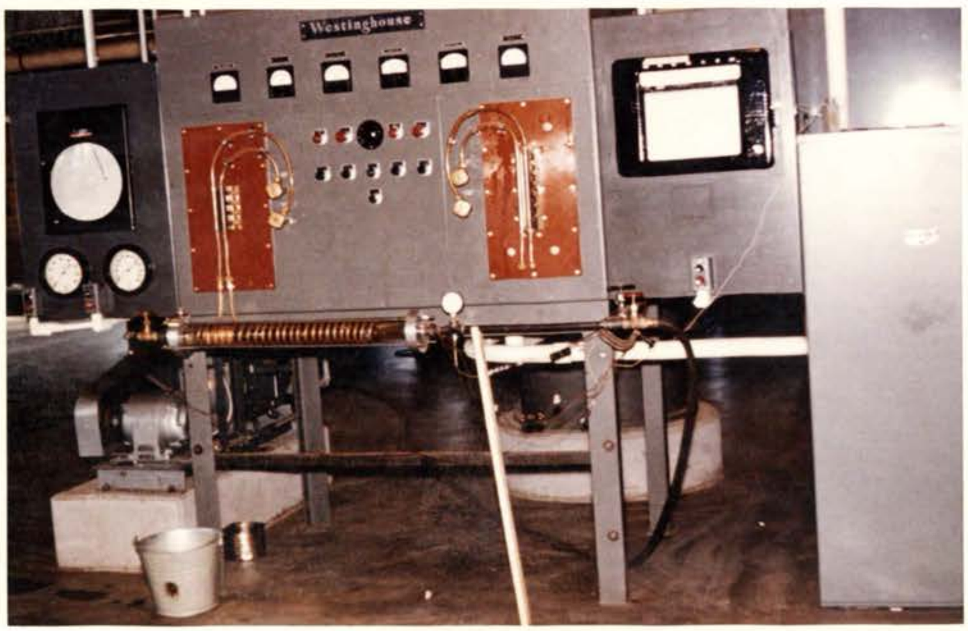
FIGURE #3




FRONT ELEVATION OF THE HIGH FREQUENCY INDUCTION HEATER

WITH WATER FLOW METER AND TEMPERATURE RECORDER

Figure No. 4




Photograph Showing the Front View of Complete Installation shown in fig. 5). The flow may be obtained inside, outside or both inside and outside of the test specimen by suitable arrangements.

(9) The heating coil was wrapped on the glass tube between points 9 and 9'. The test tube was heated by a Westinghouse multi-purpose induction heater operating with 30 KW and 10,000 cycles. It is capable of supplying voltage from 12.5 to 800 volts.

(10) A pressure gage to measure pressure at the outlet side of water.

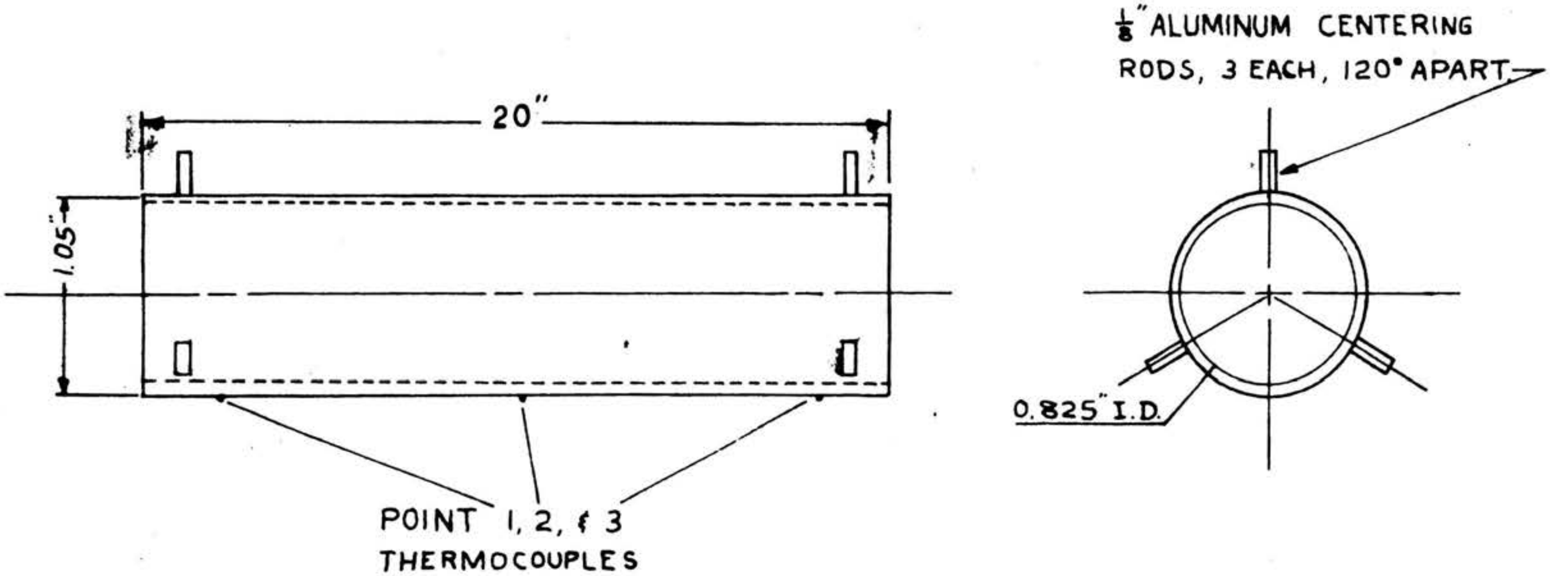
(11) A thermocouple to measure the outlet temperature of water.

(12) A throttle control valve to control and vary the water flow. It remains fully open when the heat transfer data was taken to avoid pressure fluctuations in the glass tube, which also prevents the possibility of leakage.

(13) A continuous null balance type d-c potentiometer of Barber-Colman Company, Illinois, to measure the electromotive force generated by various thermocouples and hence the temperatures of the water and test tube at different points in five second cycles.

The detailed drawings of the different test tubes have been shown in fig. 5. All the test tubes were taken from 3/4 inch nominal diameter steel pipe having I.D., 0.825" and O.D., 1.05". The length of the test tube was 20 inches. Three thermocouples were placed on each test tube, at the two ends and in the middle, to take the tube temperatures as shown in the figure.

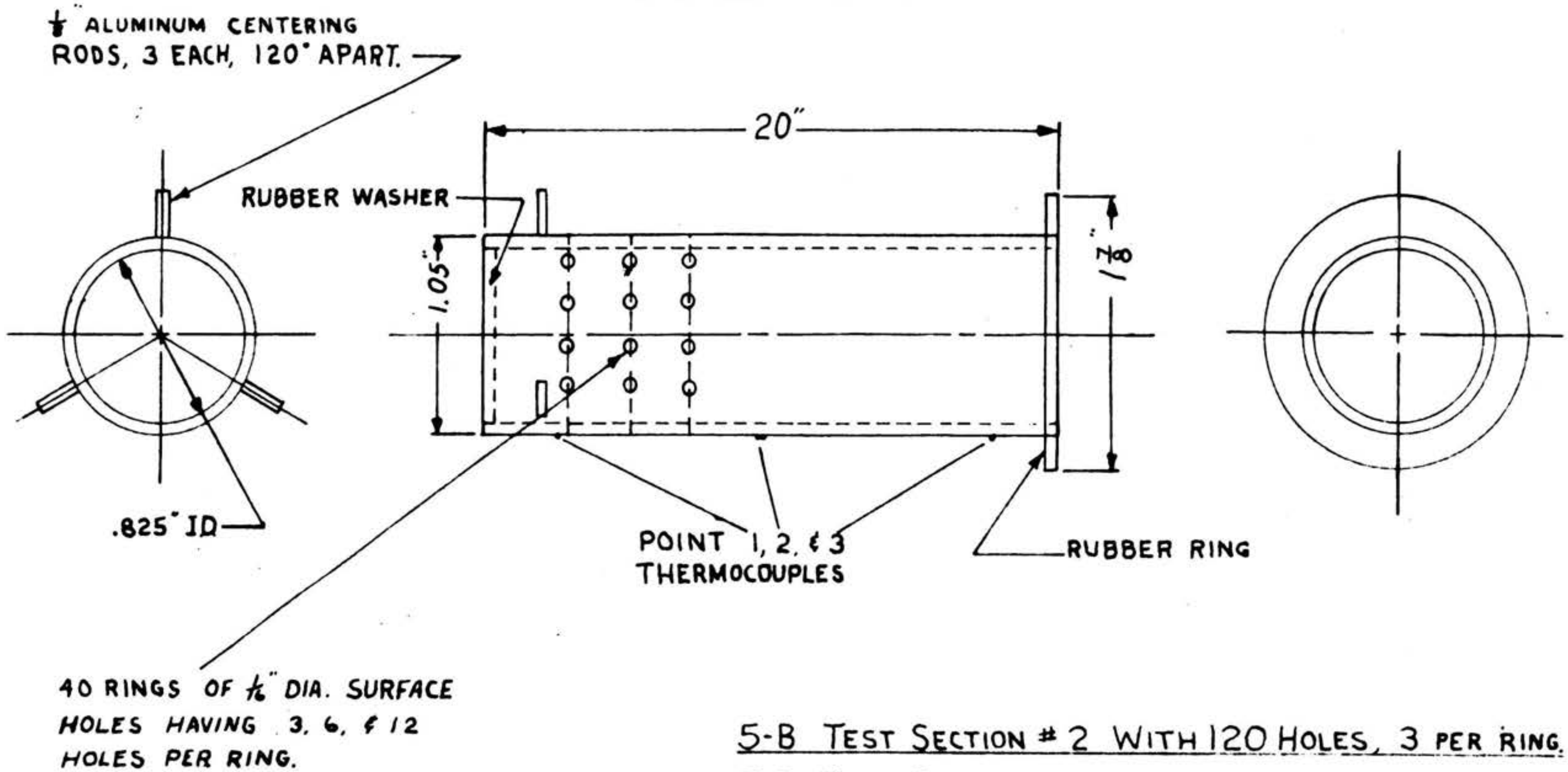
FIGURE #5-A




NOTE: DIMENSION AS SHOWN, NOT TO SCALE.

TEST SECTION NO. 1 WITHOUT SURFACE HOLES

FIGURE # 5-B,C,&D




5-B TEST SECTION # 2 WITH 120 HOLES, 3 PER RING.

5-C TEST SECTION # 3 WITH 240 HOLES, 6 PER RING.

5-D TEST SECTION # 4 WITH 480 HOLES, 12 PER RING.

NOTE: DIMENSIONS AS SHOWN, NOT TO SCALE.

The first test pipe was taken without any surface holes (fig.5a) and two data were taken on this tube. One, being both ends open, i.e. flow of water both inside and outside the pipe simultaneously, the second being flow only outside the pipe closing both ends by gluing on thick rubber pieces.

The other three test tubes were similar, having different numbers of 1/16 inch diameter holes on the surface. The holes were drilled at 1/2 inch intervals along the length of the pipe leaving 1/4 inch at both ends, thus giving 40 rings of holes in total length of 20 inches. The holes were staggered in the consecutive rings as shown in figure 5. The total number of holes was 120, 240, and 480 respectively in three test tubes.

The flow was always parallel to the length of the heated pipe. The water entered in the annular area between the pipe and the glass tube, as the inlet end of the pipe was closed. The water then entered inside the pipe through the holes and flowed out through the open end of the pipe. The annular area at the exit was closed by gluing a rubber ring, to accomplish all water flow through the open end of the pipe.

2. Experimental Procedure:

(1) Start pump.

(2) Remove all air from the glass tube by opening the air valve(6) and closing the control valve(12).

(3) Turn all power switches on panel board, potentiometer switch and flowmeter switch to "on" position.

(4) Adjust flow of water by control valve (5), opening fully the control valve (12).

(5) Adjust the power supply by powerstat on panel board to heat the pipe, as not to increase the maximum surface temperature above 200 °F.

(6) Allow the conditions to be steady.

(7) Take the required data.

(8) Vary the flow and readjust the power if required to keep the mean bulk temperature as nearly constant as possible. Take data for different water flow under steady conditions.

3. Range of Measurements.

The flow of water was varied from 800 to 1550 lbs. per hour (about 1.55 to 3.0 gallons per minute). The lower limit was chosen to keep the flow turbulent. The higher flow limit was fixed to avoid leakage at the exit end of the annular area and to obtain sufficient water temperature rise within limits of the heating capacity. The maximum pipe surface temperature was kept up to 200 °F (in the center of the pipe) to avoid water boiling at the surface.

4. Measurement and Calculating Techniques.

The following measurements were taken on the test apparatus.

(1) Rate of water flow, which was taken directly by flowmeter placed at inlet side.

(2) Inlet and Outlet water temperatures.

(3) The temperature of the test pipe surface at three points; both ends and in the center.

Some of the assumptions made in calculations are as follows.

(1) The pipe wall temperature was same at outside and inside surfaces.

(2) The temperature distribution along the pipe length was taken as linear, which was highest at the central point.

(3) The average wall temperature was calculated by measuring the areas under the longitudinal temperature distribution curves (linear) and dividing by the heat transfer length.

(4) Steady state conditions.

(5) Bulk temperature was considered to be thoroughly mixed water temperature at any cross section.

(6) The average bulk temperature was taken as the mean of inlet and outlet water temperatures.

(7) Temperature loss at the throttle valve was neglected.

(8) Fluid does no work.

(9) Heat transfer from 1/8 inch diameter aluminum center rods fixed at the test-section ends was not taken into account.

(10) Heat transfer from two end thicknesses of the pipe was neglected.

(11) Heat loss from water to flow pipe and outside between the points of measuring inlet and outlet water temperatures was assumed as very small.

(12) The specific heat of water was taken as constant within this temperature range equal to 1 Btu per lb. $^{\circ}$F.

(13) The value of film coefficient 'h' was assumed to be constant along the pipe length and so the average value was calculated.

(14) The change of heat transfer area due to 1/16 inch diameter holes was taken into account.

The following procedure was adopted for calculations showing the results.

(1) $T = \frac{T_1 + T_2}{2}$

(2) $t = \frac{\text{temp. point 1} + 2 \times \text{temp. point 2} + \text{temp. point 3}}{4}$

(3) $\Delta t = T-t$

(4) $q = -h A \Delta t = W C_p (T_2 - T_1)$

or $h = -\frac{W (T_2 - T_1)}{\Delta t \times A}$ taking cp = 1

The value of 'h' was calculated from the above equation. The graph was drawn between W and h on logarithmic coordinates taking 'h' as ordinate and 'W' as abscissa.

5. Experimental Results.

The experimental data and results have been shown in Table No. 1 to 5. Two tests were performed on every test pipe to get more accuracy. These two test results have been shown in tables under (a) and (b) respectively. It was noted that the results of the two different tests were quite comparable. The graphs of film coefficient h, vs mass flow W have been shown separately for each test pipe and then a combined graph has also been shown(figure no. 11) for comparison. In drawing graphs all readings calculated in two different tests on the same test pipe were plotted and a mean curve was drawn.

TABLE NO. 1(a)

HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20"LONG: A=0.817 Sq. ft.

both ends open, without surface holes : fig. 5(a)

TEST NO. I

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h $\frac{Btu}{hr.ft.^2 °F}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 120.0 | 132.0 | 135 | 186 | 172 | 169.75 | 126.0 | 12.0 | 43.75 | 270 |
| 2 | 75 | 968 | 120.0 | 131.5 | 131 | 185 | 165 | 166.5 | 125.75 | 11.5 | 40.75 | 335 |
| 3 | 100 | 1108 | 120.0 | 132.0 | 132 | 185 | 165 | 166.75 | 126.0 | 12.0 | 40.75 | 398 |
| 4 | 125 | 1220 | 120.0 | 132.5 | 133 | 185 | 164 | 166.75 | 126.25 | 12.5 | 40.5 | 460 |
| 5 | 150 | 1330 | 121.0 | 133.5 | 133 | 185 | 164 | 166.75 | 127.25 | 12.5 | 39.5 | 515 |
| 6 | 175 | 1435 | 121.5 | 133.5 | 132 | 183 | 162 | 165.0 | 127.5 | 12.0 | 37.5 | 562 |
| 7 | 200 | 1540 | 121.0 | 133.5 | 134 | 186 | 156 | 165.5 | 127.25 | 12.5 | 38.25 | 615 |

TABLE NO. 1(b)

HEAT TRANSFER EXPERIMENTAL DATA

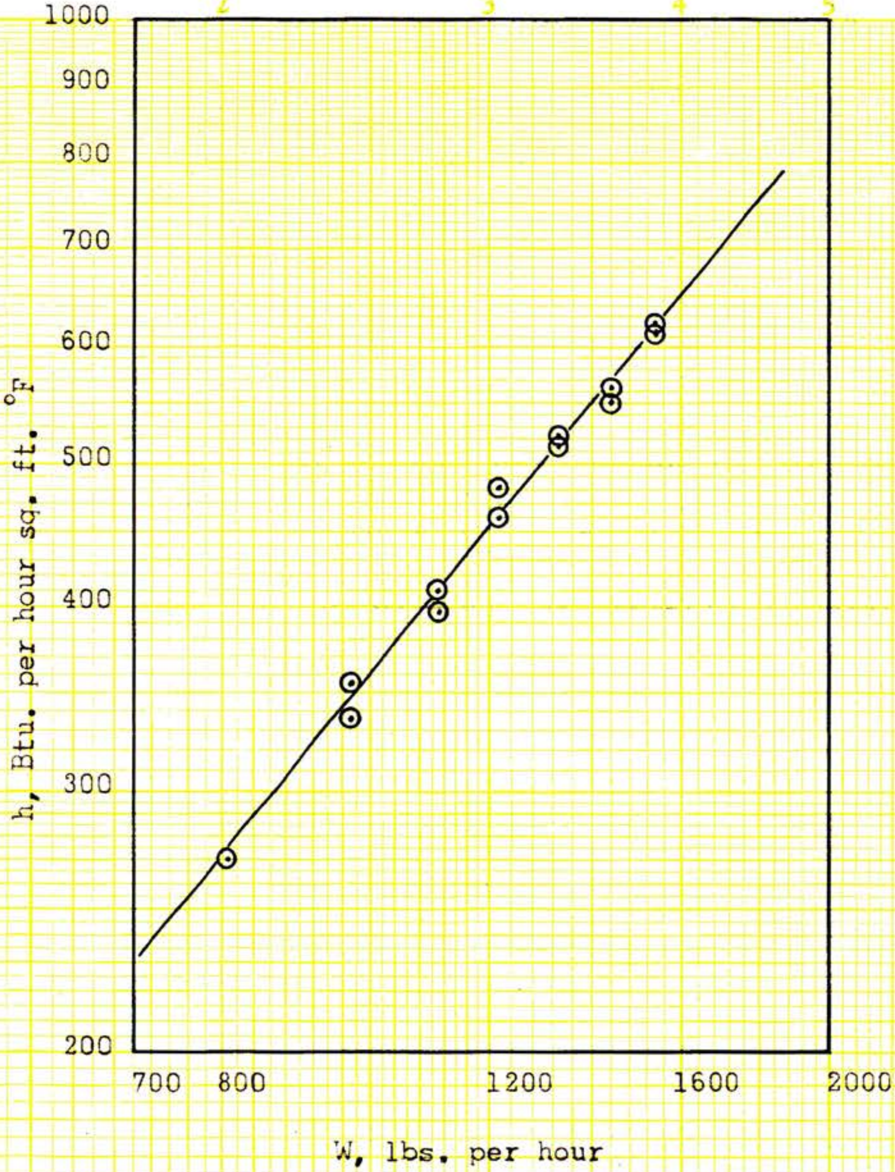
TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.817 Sq. ft.

both ends open, without surface holes : fig. 5(a)

TEST NO. II

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h $\frac{\text{Btu}}{\text{hr.ft.}^2\text{°F}}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 116 | 130.0 | 140 | 190 | 175 | 173.75 | 123.0 | 14.0 | 50.75 | 272 |
| 2 | 75 | 968 | 116 | 130.0 | 135 | 187 | 169 | 169.5 | 123.0 | 14.0 | 46.5 | 357 |
| 3 | 100 | 1108 | 116 | 129.0 | 134 | 183 | 162 | 165.5 | 122.5 | 13.0 | 43.0 | 410 |
| 4 | 125 | 1220 | 116 | 129.0 | 132 | 180 | 160 | 163.0 | 122.5 | 13.0 | 40.5 | 480 |
| 5 | 150 | 1330 | 116 | 130.0 | 134 | 185 | 163 | 166.75 | 123.0 | 14.0 | 43.75 | 520 |
| 6 | 175 | 1435 | 116 | 130.5 | 135 | 189 | 166 | 169.75 | 123.25 | 14.5 | 46.5 | 550 |
| 7 | 200 | 1540 | 115 | 129.5 | 131 | 185 | 164 | 166.25 | 122.25 | 14.5 | 44.0 | 620 |

Figure no. 6




Film Coefficient, vs Mass Flow for Test Section No. I, without surface holes and both ends open

TABLE NO. 2(a)

HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.457 Sq. ft.

both ends closed, without surface holes : fig. 5(a)

TEST NO. I

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h $\frac{Btu}{hr.ft.^2 °F}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 119.0 | 129.0 | 136 | 185 | 158 | 166.0 | 124 | 10 | 42.0 | 420 |
| 2 | 75 | 968 | 120.5 | 129.5 | 134 | 181 | 156 | 163.0 | 125 | 9 | 38.0 | 500 |
| 3 | 100 | 1108 | 119.0 | 129.0 | 131 | 187 | 157 | 165.5 | 124 | 10 | 41.5 | 583 |
| 4 | 125 | 1220 | 119.0 | 129.0 | 131 | 187 | 157 | 165.5 | 124 | 10 | 41.5 | 644 |
| 5 | 150 | 1330 | 119.0 | 129.0 | 130 | 187 | 157 | 165.25 | 124 | 10 | 41.25 | 706 |
| 6 | 175 | 1435 | 119.0 | 129.0 | 132 | 184 | 160 | 165.0 | 124 | 10 | 41.0 | 764 |
| 7 | 200 | 1540 | 120.0 | 130.0 | 132 | 187 | 158 | 166.0 | 125 | 10 | 41.0 | 823 |

TABLE NO. 2(b)

HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.457 Sq. ft.

both ends closed, without surface holes : fig. 5(a)

TEST NO. II

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h $\frac{\text{Btu}}{\text{hr.ft.}^2\,°\text{F}}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 121.0 | 131.5 | 132.0 | 190 | 170 | 170.5 | 126.25 | 10.5 | 44.25 | 420 |
| 2 | 75 | 968 | 120.5 | 131.0 | 132.5 | 190 | 165 | 169.37 | 125.75 | 10.5 | 43.62 | 510 |
| 3 | 100 | 1108 | 120.5 | 130.5 | 131.0 | 186 | 165 | 167.0 | 125.5 | 10.0 | 41.5 | 583 |
| 4 | 125 | 1220 | 120.5 | 130.5 | 131.0 | 186 | 167 | 167.5 | 125.5 | 10.0 | 42.0 | 635 |
| 5 | 150 | 1330 | 121.0 | 131.0 | 131.0 | 185 | 165 | 166.5 | 126.0 | 10.0 | 40.5 | 720 |
| 6 | 175 | 1435 | 121.0 | 132.0 | 133.0 | 190 | 170 | 170.75 | 126.5 | 11.0 | 44.25 | 780 |
| 7 | 200 | 1540 | 121.0 | 132.5 | 135.0 | 193 | 175 | 174.0 | 126.75 | 11.5 | 47.25 | 820 |

Figure no. 7


Film Coefficient, vs Mass Flow for Test Section

No. I, without surface holes and both ends closed

TABLE NO. 3(a)

HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.83 Sq. ft.

one end closed, 120 nos. 1/16" dia. surface holes: fig.5(b)

TEST NO. I

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h Btu/hr.ft.$^2$°F |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 122.0 | 146.0 | 144 | 191 | 175 | 175.25 | 134.0 | 24.0 | 41.25 | 565 |
| 2 | 75 | 968 | 121.0 | 142.5 | 140 | 187 | 171 | 171.25 | 131.75 | 21.5 | 39.5 | 635 |
| 3 | 100 | 1108 | 125.0 | 146.5 | 145 | 195 | 177 | 178.0 | 135.75 | 21.5 | 42.25 | 680 |
| 4 | 125 | 1220 | 124.5 | 144.5 | 143 | 190 | 175 | 174.5 | 134.5 | 20.0 | 40.0 | 735 |
| 5 | 150 | 1330 | 125.0 | 144.5 | 144 | 190 | 175 | 174.75 | 134.75 | 19.5 | 40.0 | 780 |
| 6 | 175 | 1435 | 125.0 | 143.5 | 142 | 187 | 175 | 172.75 | 134.25 | 18.5 | 38.5 | 830 |
| 7 | 200 | 1540 | 124.5 | 145.0 | 145 | 192 | 181 | 177.5 | 134.75 | 20.5 | 42.75 | 890 |

TABLE NO. 3(b)

HEAT TRANSFER EXPERIMENTAL DATA

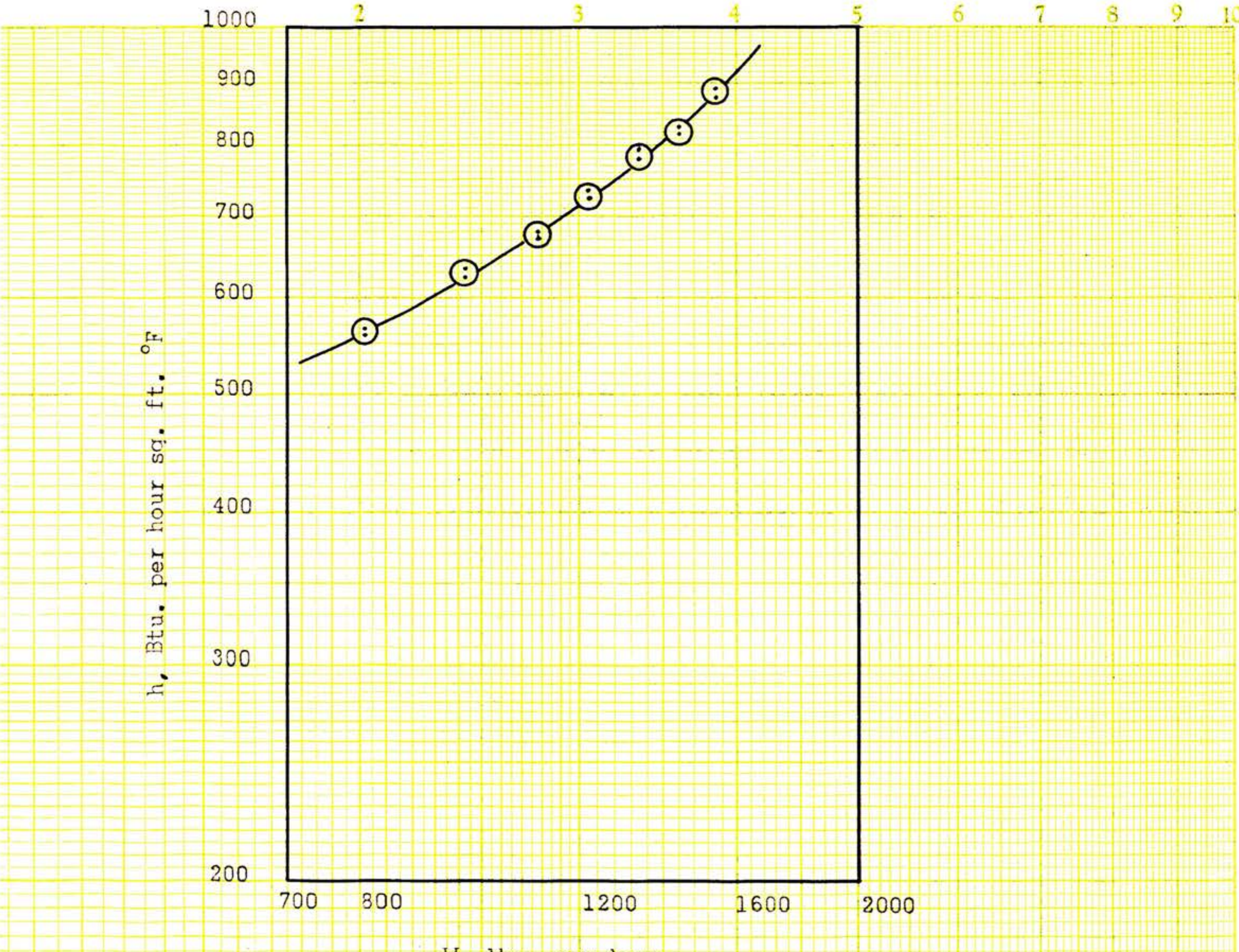
TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.83 Sq. ft.

one end closed, 120 nos. 1/16" dia. surface holes : fig. 5(b)

TEST NO. II

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h Btu / hr.ft.$^2$°F |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 127.5 | 149.5 | 147 | 192 | 175 | 176.5 | 138.5 | 22.0 | 38.0 | 560 |
| 2 | 75 | 968 | 126.5 | 149.0 | 146 | 197 | 179 | 179.75 | 137.75 | 22.5 | 42.0 | 625 |
| 3 | 100 | 1108 | 128.5 | 147.5 | 145 | 191 | 175 | 175.5 | 138.0 | 19.0 | 37.5 | 676 |
| 4 | 125 | 1220 | 129.5 | 147.5 | 145 | 190 | 175 | 175.0 | 138.5 | 18.0 | 36.5 | 725 |
| 5 | 150 | 1330 | 130.0 | 149.0 | 145 | 195 | 176 | 177.75 | 139.5 | 19.0 | 38.25 | 795 |
| 6 | 175 | 1435 | 130.0 | 148.5 | 145 | 195 | 178 | 178.25 | 139.25 | 18.5 | 39.0 | 824 |
| 7 | 200 | 1540 | 130.0 | 148.5 | 145 | 195 | 179 | 178.5 | 139.25 | 18.5 | 39.25 | 877 |

Figure no. 8




Film Coefficient, vs Mass Flow for Test Section No. 2

with 120 nos. surface holes and inlet end closed

TABLE NO. 4(a)

HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825 ", I.D.; 1.05", O.D.; 20" LONG: A=0.8436 Sq. ft.

one end closed, 240 nos. 1/16" dia. surface holes : fig. 5(c)

TEST NO. I

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h Btu/hr.ft.$^2$°F |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 116.0 | 139.0 | 140 | 185 | 170 | 170.0 | 127.5 | 23.0 | 42.5 | 517 |
| 2 | 75 | 968 | 117.5 | 134.5 | 136 | 174 | 164 | 162.0 | 126.0 | 17.0 | 36.0 | 545 |
| 3 | 100 | 1108 | 117.5 | 136.0 | 140 | 179 | 174 | 168.0 | 126.75 | 18.5 | 41.25 | 590 |
| 4 | 125 | 1220 | 119.0 | 136.5 | 140 | 180 | 174 | 168.5 | 127.75 | 17.5 | 40.75 | 620 |
| 5 | 150 | 1330 | 119.0 | 135.0 | 137 | 176 | 170 | 164.75 | 127.0 | 16.0 | 37.75 | 670 |
| 6 | 175 | 1435 | 120.0 | 136.0 | 137 | 178 | 171 | 166.0 | 128.0 | 16.0 | 38.0 | 718 |
| 7 | 200 | 1540 | 119.5 | 136.5 | 139 | 179 | 174 | 167.75 | 128.0 | 17.0 | 39.75 | 780 |

TABLE NO. 4(b)

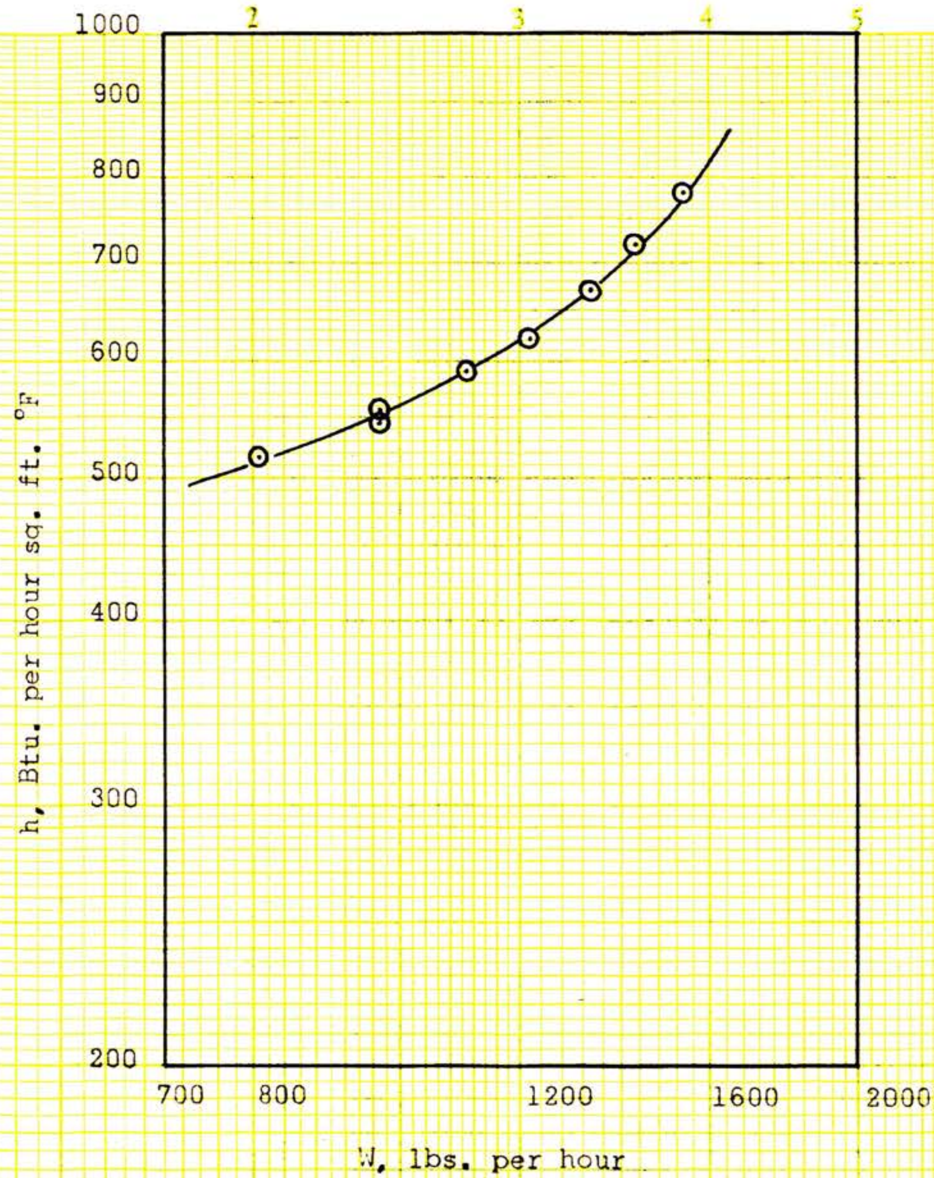
HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.8436 Sq. ft.

one end closed, 240 nos. 1/16" dia. surface holes : fig. 5(c)

| RUN NO. | FLOW METER READING | W lbs/hr | WATER TEMP °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $\sim\Delta t$ °F | h $\frac{\text{Btu}}{\text{hr.ft.}^2\text{°F}}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 124.0 | 147.0 | 149 | 192 | 180 | 178.25 | 135.5 | 23.0 | 42.75 | 514 |
| 2 | 75 | 968 | 124.0 | 145.0 | 149 | 190 | 183 | 178.0 | 134.5 | 21.0 | 43.5 | 555 |
| 3 | 100 | 1108 | 125.5 | 145.5 | 150 | 192 | 185 | 179.75 | 135.5 | 20.0 | 44.25 | 590 |
| 4 | 125 | 1220 | 125.5 | 144.5 | 150 | 191 | 184 | 179.0 | 135.0 | 19.0 | 44.0 | 623 |
| 5 | 150 | 1330 | 124.5 | 143.0 | 150 | 190 | 180 | 177.5 | 133.75 | 18.5 | 43.75 | 670 |
| 6 | 175 | 1435 | 124.5 | 143.5 | 150 | 190 | 186 | 179.0 | 134.0 | 19.0 | 45.0 | 720 |
| 7 | 200 | 1540 | 124.0 | 142.0 | 148 | 186 | 180 | 175.0 | 133.0 | 18.0 | 42.0 | 782 |

Figure no. 9




Film Coefficient, vs Mass Flow for Test Section No. 3

with 240 nos. surface holes and inlet end closed

TABLE NO. 5(a)

HEAT TRANSFER EXPERIMENTAL DATA

TEST SECTION:- STEEL PIPE 0.825", I.D.; 1.05", O.D.; 20" LONG: A=0.87 Sq. ft.

one end closed, 480 nos. 1/16" dia. surface holes : fig. 5(d)

TEST NO. I

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F | | SURFACE TEMPERATURE °F | | | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h $\frac{\text{Btu}}{\text{hr.ft.}^2\,°\text{F}}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $T_1$ | $T_2$ | POINT 1 | POINT 2 | POINT 3 | | | | | |
| 1 | 50 | 805 | 127.0 | 155.0 | 173 | 186 | 177 | 180.5 | 141.0 | 28.0 | 39.5 | 656 |
| 2 | 75 | 968 | 128.0 | 151.5 | 167 | 184 | 174 | 177.25 | 139.75 | 23.5 | 37.5 | 696 |
| 3 | 100 | 1108 | 128.0 | 151.0 | 170 | 185 | 176 | 179.0 | 139.5 | 23.0 | 39.5 | 743 |
| 4 | 125 | 1220 | 128.5 | 149.5 | 167 | 182 | 175 | 176.5 | 139.0 | 21.0 | 37.5 | 785 |
| 5 | 150 | 1330 | 130.0 | 150.5 | 167 | 185 | 177 | 178.5 | 140.25 | 20.5 | 38.25 | 820 |
| 6 | 175 | 1435 | 130.0 | 151.0 | 171 | 186 | 178 | 180.25 | 140.5 | 21.0 | 39.75 | 870 |
| 7 | 200 | 1540 | 130.0 | 152.0 | 174 | 189 | 180 | 183.0 | 141.0 | 22.0 | 42.0 | 925 |

TABLE NO. 5(b)

HEAT TRANSFER EXPERIMENTAL DATA

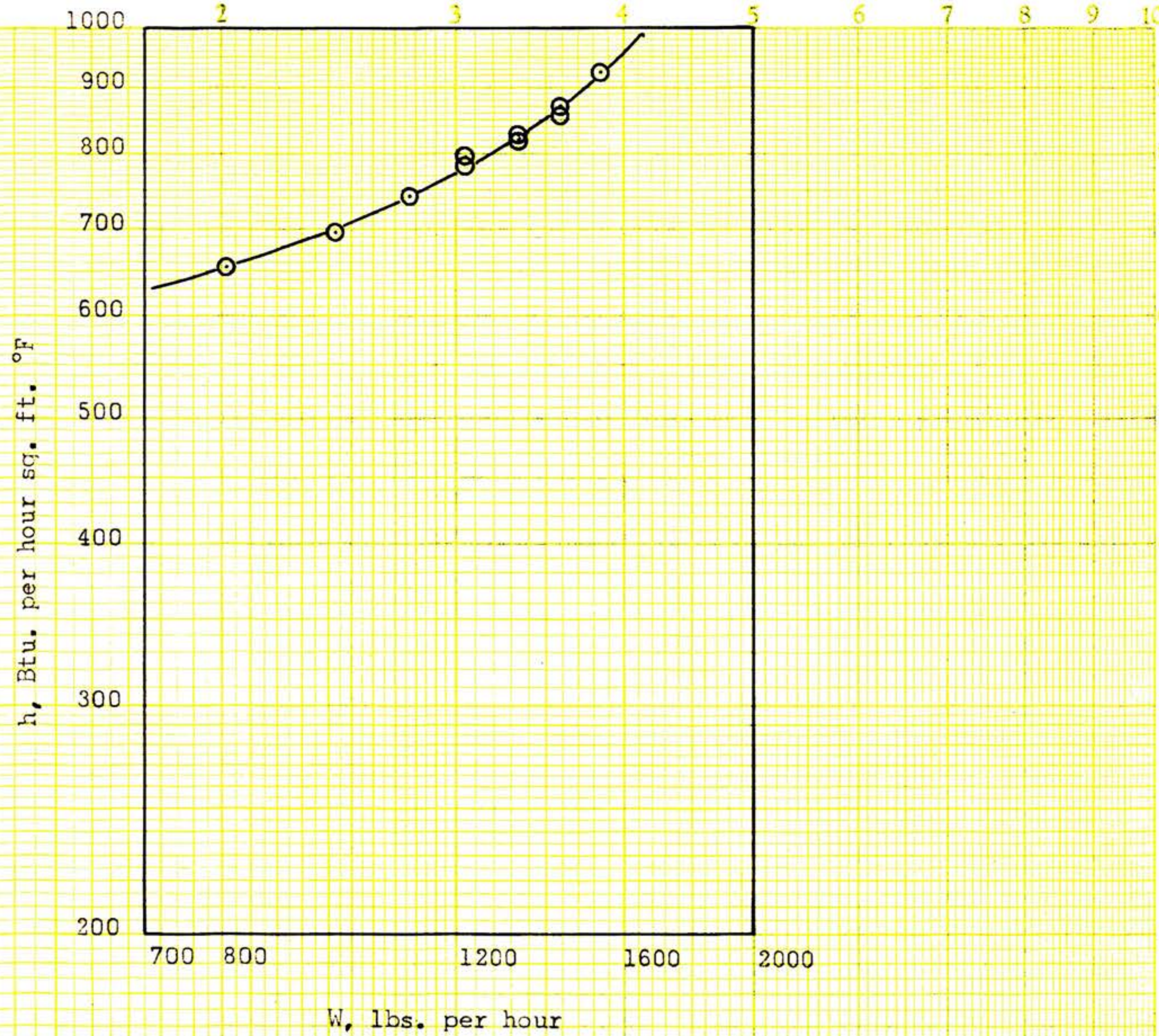
TEST SECTION:- STEEL PIPE, 0.825", I.D.; 1.05", O.D.; 20" LONG: **A=0.87 Sq. ft.**

**one end closed, 480 nos. 1/16" dia. surface holes : fig. 5(d)**

TEST NO. II

| RUN NO. | FLOWMETER READING | W lbs/hr. | WATER TEMP. °F $T_1$ | $T_2$ | SURFACE TEMPERATURE °F POINT 1 | POINT 2 | POINT 3 | t °F | T °F | $(T_1-T_2)$ °F | $-\Delta t$ °F | h Btu / hr.ft.$^2$°F |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 50 | 805 | 121.5 | 151.0 | 167 | 185 | 175 | 178.0 | 135.25 | 29.5 | 41.75 | 654 |
| 2 | 75 | 968 | 122.0 | 147.0 | 165 | 180 | 172 | 174.25 | 134.5 | 25.0 | 39.75 | 700 |
| 3 | 100 | 1108 | 124.0 | 150.0 | 173 | 189 | 175 | 181.5 | 137.0 | 26.0 | 44.5 | 742 |
| 4 | 125 | 1220 | 125.0 | 148.5 | 165 | 186 | 175 | 178.0 | 136.75 | 23.5 | 41.25 | 795 |
| 5 | 150 | 1330 | 124.5 | 147.5 | 170 | 185 | 174 | 178.5 | 136.0 | 23.0 | 42.5 | 826 |
| 6 | 175 | 1435 | 125.5 | 147.5 | 170 | 185 | 175 | 178.75 | 136.5 | 22.0 | 42.25 | 860 |
| 7 | 200 | 1540 | 127.0 | 149.0 | 173 | 185 | 177 | 180.0 | 138.0 | 22.0 | 42.0 | 925 |

Figure no. 10




Film Coefficient, vs Mass Flow for Test Section No. 4

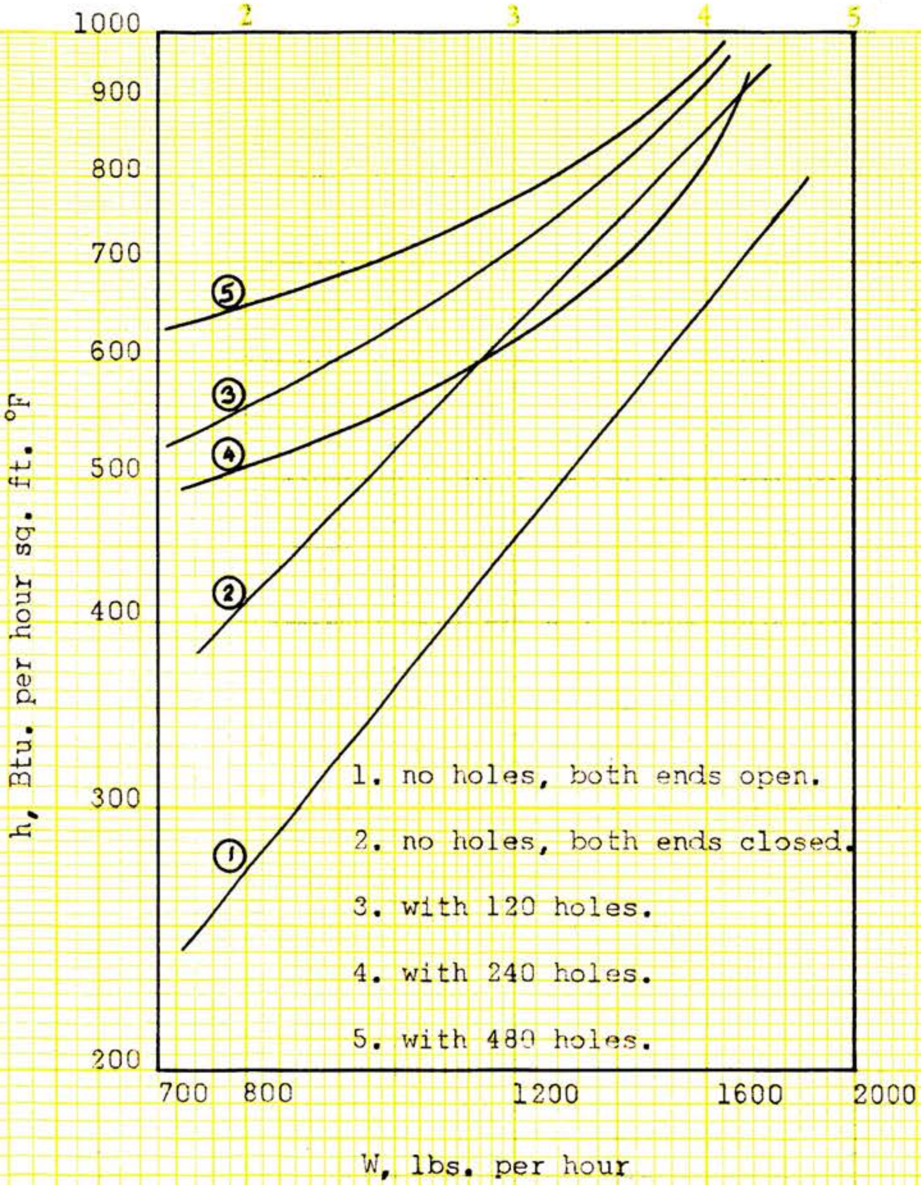
with 480 nos. surface holes and inlet end closed

TABLE NO. 6

HEAT TRANSFER COEFFICIENTS FOR VARIOUS TEST SECTIONS

| WATER FLOW lbs. per hour | COEFFICIENT OF CONVECTIVE HEAT TRANSFER (h), Btu per hr. ft$^2$°F | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | TEST SECTION 1 flow, both inside and outside | | TEST SECTION 1 flow outside only | | TEST SECTION 2 with 120 holes | | TEST SECTION 3 with 240 holes | | TEST SECTION 4 with 480 holes | |
| | TEST I | TEST II | TEST I | TEST II | TEST I | TEST II | TEST I | TEST II | TEST I | TEST II |
| 805 | 270 | 272 | 420 | 420 | 565 | 560 | 517 | 514 | 656 | 654 |
| 968 | 335 | 357 | 500 | 510 | 635 | 625 | 545 | 555 | 696 | 700 |
| 408 | 398 | 410 | 583 | 583 | 680 | 676 | 590 | 590 | 743 | 742 |
| 1220 | 460 | 480 | 644 | 635 | 735 | 725 | 620 | 623 | 785 | 795 |
| 1330 | 515 | 520 | 706 | 720 | 780 | 795 | 670 | 670 | 820 | 826 |
| 1435 | 562 | 550 | 764 | 780 | 830 | 824 | 718 | 720 | 870 | 860 |
| 1540 | 615 | 620 | 823 | 820 | 890 | 877 | 780 | 782 | 925 | 925 |

Figure no. 11




Film Coefficient, vs Mass Flow for all

Test Sections Combined for Comparison

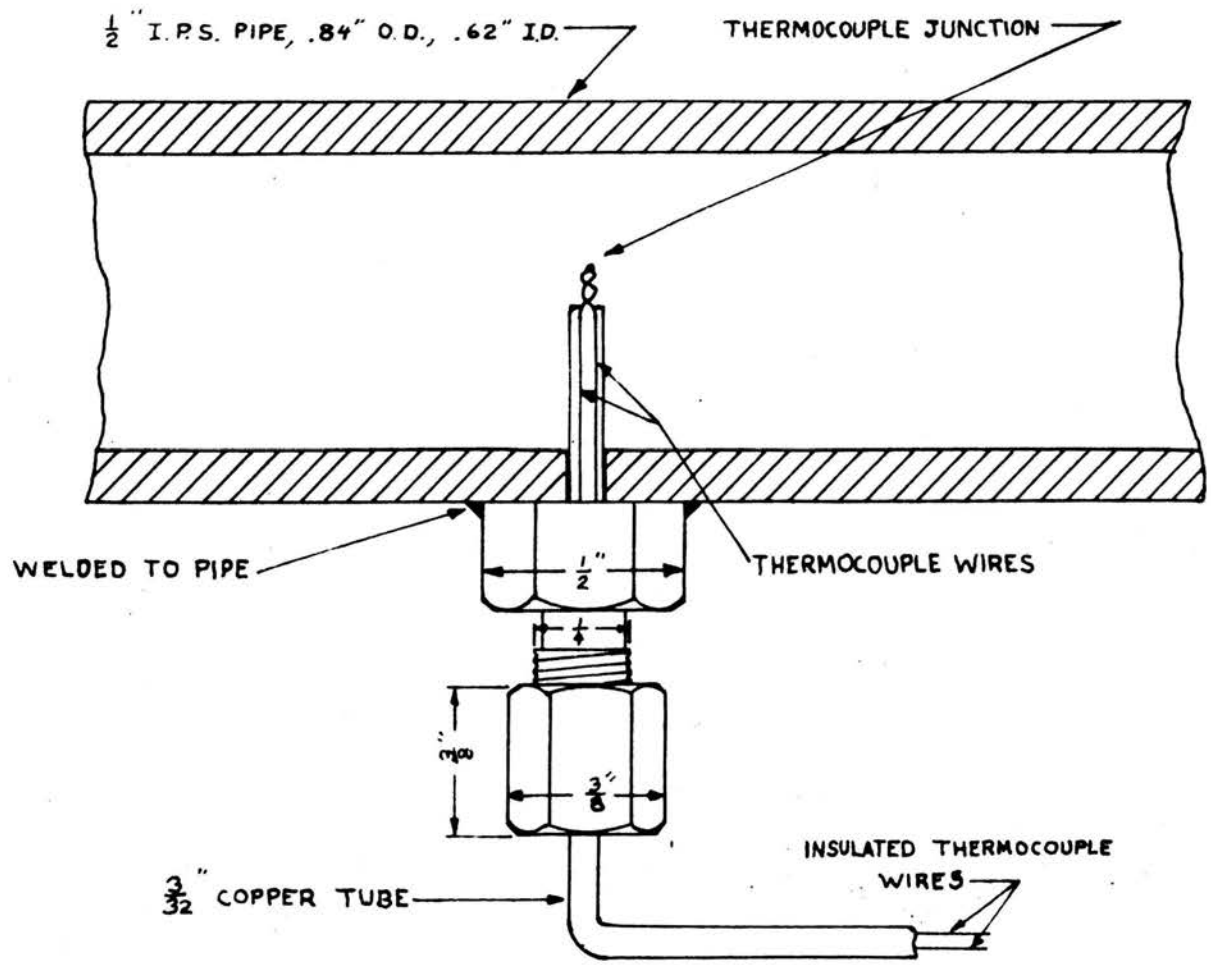
6. Accuracy of Measurements.

The accuracy of measurements of heat transfer coefficient was limited by several factors. As only the comparison of two heat transfer coefficients, with and without boundary layer control, has been given in this volume and this investigation has been restricted to qualitative results rather than strict quantitative, many sources of error were neutralized which have the same effect on the heat transfer coefficient in two cases. Some of the sources of error are as follows.

Temperatures were measured by iron-constantan thermocouples attached at three points on the test pipe and in the inlet and outlet water pipes. The thermal electromotive forces of thermocouples were recorded by a continuous null balance type d-c potentiometer, whose sensitivity and accuracy was $\pm$ 1°F. As the water temperature rise was quite low because of maximum surface temperature limit, the error of even $\pm$ 1°F in water temperatures may introduce an error of 5 to 10% in the results.

The largest source of error in temperature measurements arises from deviation of thermocouple temperature from the true temperature to be measured. The wall temperatures were determined by averaging the readings of various thermocouples located on the test pipe. The wall temperature of the inside and outside surface was taken as same.

FIGURE # 12




THERMOCOUPLE INSTALLATION TO MEASURE WATER TEMPERATURE IN PIPE

The water flow was measured by Honeywell's "pressure differential" type flowmeter, which was calibrated under the same conditions as the actual experiment. Its calibration accuracy was about $\pm$ 1%, which would not affect the results appreciably.

7. Discussion of Results.

The experiments performed and the results obtained for the convective heat transfer coefficient with and without boundary layer control are only accurate within $\pm$5%, due to the accuracy of temperature measurements within $\pm$ 1°F., and the small rise in water temperatures.

The results with boundary layer control are compared to the results obtained by flowing water both on the outside and inside surfaces of the pipe without any boundary layer control. The values of the heat transfer coefficient have also been obtained with water flow on the outside surface only, but the heat transfer area in this case is much less, as the flow is only on one side, and the rate of heat transfer is much less in spite of a slight increase in 'h'. In actual applications, if this type of boundary layer control is applicable, the flow on both sides of the surface can very well be obtained, so this comparison seems to be more applicable to actual problems.

The effect of boundary layer control by drilling 1/16 inch diameter holes on the pipe surface is to increase the convective heat transfer coefficient, and hence the heat transfer rate per unit surface area. The values of the heat transfer coefficient are 50 to

140% higher with boundary layer control, keeping the same water flow in two cases. The percentage increase of the heat transfer coefficient is a function of the number of surface holes.

Fig. 6 shows the values of 'h' versus 'W' without any boundary layer control and with the flow both outside and inside of the test section, which is a straight line on logarithmic coordinates and is in agreement with the general heat transfer coefficient equation (3). The straight line shown in fig. 7 also follows equation (3) for the flow on the outside of the pipe only.

It is clear from figures 8, 9, & 10 that the values of 'h' vs. 'W' with boundary layer control on logarithmic coordinates are not straight lines, and so the equation (3) can not be applied in such cases.

It has been found by calculations that, if the quantity of water entering through each hole is assumed to be equal, the flow is stream-line for about 4 inches in length at the inside surface of the inlet end, and for 5 inches in length at the outside surface of the outlet end, for the lowest rate of water flow (1.6 g.p.m. ). The flow is stream-line on certain portions of the outside and inside surfaces for all values of flow within this range (1.6 to 3.0 g.p.m.). The flow through various holes is stream-line also. This stream-line flow combined with turbulent flow at the remaining parts of the surface gives the curves between 'h' and 'W'.

Fig. 11 shows all curves for comparing the various results. It can be seen from this figure that the percentage increase in the coefficient of convective heat transfer is highest for 480 holes whose area is 2.8 times the cross-sectional area of the pipe. The effect of boundary layer control is minimum with 240 holes. It is also clear from the graphs that the boundary layer control has a greater effect at low rates of flow.

This comparison of the results shows that the minimum increase in the heat transfer coefficient should be somewhere between 240 and 120 holes. It seems that as the total area of the holes drilled approaches the cross-sectional area of the pipe, the coefficient of heat transfer decreases. It is also obvious that if the number of holes per unit length of pipe is insufficient, the effect of the boundary layer control will be small and more power will be needed to force the water through the holes. The heat transfer is greater when the total area of holes is 0.7 times the cross-sectional area of the pipe than when it is 1.4 times. The value of the heat transfer coefficient is much higher when this area ratio of the holes to the cross-sectional area of the pipe is 2.8 and the total number of holes is 480. It shows that the heat transfer coefficient increases with the increase in number of holes above 240.

One disadvantage of this type of boundary layer control lies in the fact that it can only be applied to increase the heat transfer where the heat is generated internally in the pipe material. In most of the cases the internal heat generated is proportional to the quantity of the material. By drilling holes the quantity of the material will reduce and the internal heat generated will be less. By making a larger number of holes there will be an appreciable effect on the internal heat generated which may decrease the efficiency of the system.

At this stage it is not possible to discuss the results of heat transfer and the nature of the curves obtained with boundary layer control, unless the effects of drilling holes at the pipe surface on the various characteristics of the boundary layer are known in more detail. In view of the present literature available for boundary layer control of compressible fluids, it is expected that this type of boundary layer control for incompressible fluids may have the following effects on the boundary layer characteristics and hence on the heat transfer coefficient.

(1) The thickness of the stagnation portion of the boundary layer will reduce, as some motion will be created in the fluid adjacent to the solid wall due to the water flowing through the holes. The effect of this reduction in thickness will necessarily be to increase the heat transfer coefficient.

(2) The Reynold's number on the outside of the pipe will be reduced, which will decrease the heat transfer coefficient on the outside surface.

(3) The water entering the pipe through a number of holes

will join the main water stream in a perpendicular direction, which will create more turbulence inside the pipe and hence increase the heat transfer coefficient at the inside surface.

The overall change in the heat transfer coefficient with different numbers of holes will largely depend on which of these above effects are predominating, and to what extent.

It may be noted by this discussion that the value of the heat transfer coefficient will be different at different sections and at the outside and inside surfaces of the pipe due to the different Reynold's number. Here only the average value of the heat transfer coefficient has been calculated.

# V. CONCLUSIONS

The experiments performed and the results obtained for heat transfer with and without boundary layer control conclude the following:

1. The effect of boundary layer control through 1/16 inch diameter surface holes is to increase the heat transfer coefficient and hence the heat transfer rate from pipe surface to water per unit surface area.
2. The total heat transfer area increases slightly due to the drilling of holes, which is also a factor for increasing the heat transfer from the same pipe.
3. The percentage increase in the coefficient of heat transfer is higher at low rates of water flow.
4. The highest increase in the heat transfer coefficient is approximately 140% at lower flow rates of 1.6 gallons of water per minute, and 50% at higher flow rates of 3.0 gallons per minute, compared with the values obtained with flow on both the outside and the inside surfaces of the pipe simultaneously.
5. The highest increase in the heat transfer coefficient can be obtained by drilling 480 holes, whose total area is 2.8 times the cross sectional area of the pipe. The range of increase is 50% to 140%.
6. The lowest increase in the heat transfer coefficient was given by 240 holes whose total area was 1.4 times the cross sectional area of the pipe. The range of increase is 25% to 50%.
7. When the total area of the surface holes is decreased to about 0.7 times the cross sectional area, the coefficient of heat transfer

increases that of values obtained by 1.4 times the area of the holes, but is less than the values given by 2.8 times the area of the holes. The range of increase is 42% to 70%.

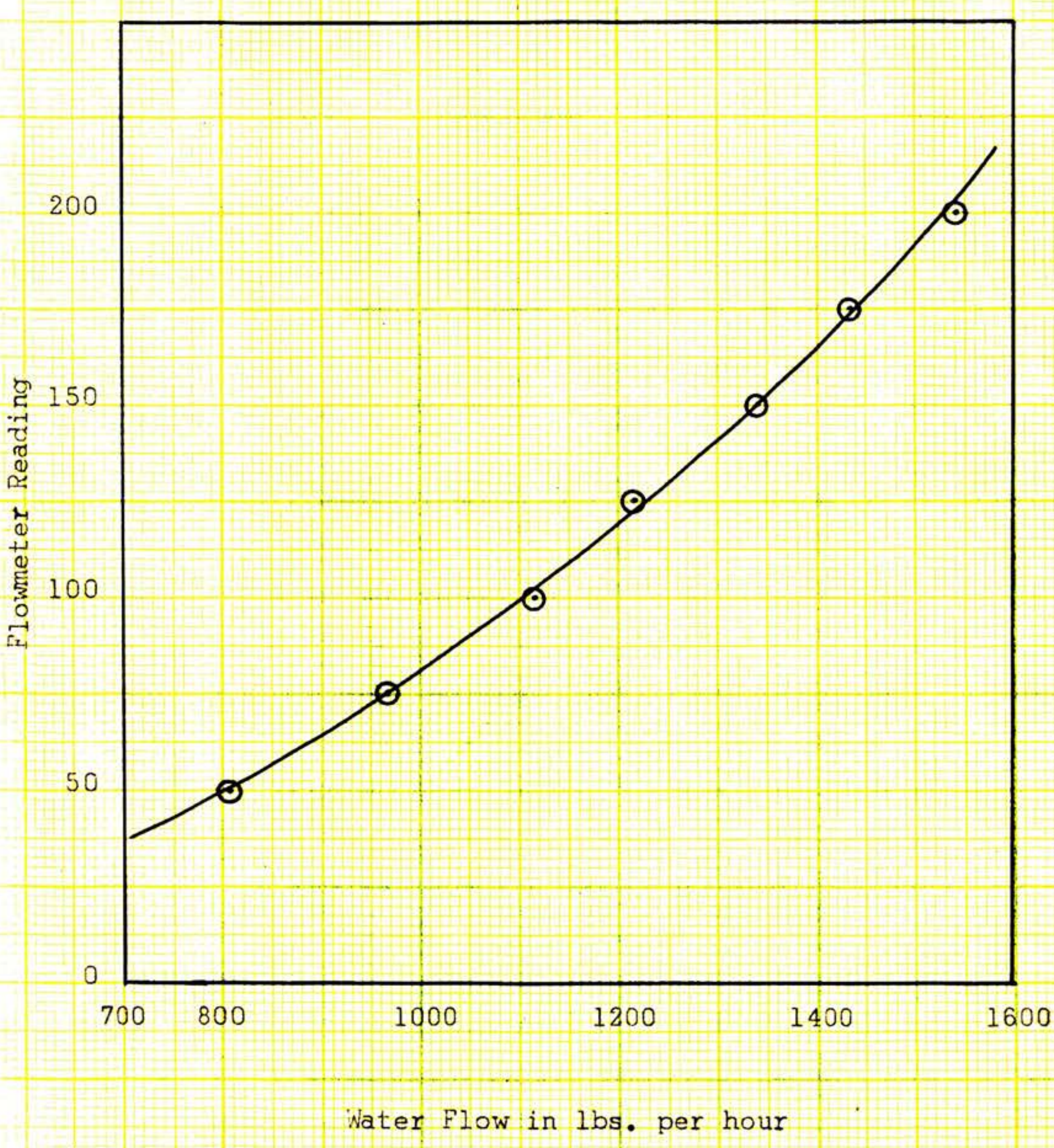
8. The curves between 'h' and 'W' on logarithmic coordinates are not straight line relations with boundary layer control, as it is without such control. These curves do not follow the general equation (3) for the coefficient of convective heat transfer.

# VI. SUMMARY

The heat transfer data was taken on test pipes of 0.825", I.D.; 1.05", O.D.; and 20 inches long with and without boundary layer control. The boundary layer control was obtained by drilling 1/16 inch diameter holes on the pipe surface. Three different test pipes with boundary layer control were tested for heat transfer with 120, 240, and 480 holes respectively. The flow was varied from 1.6 to 3.0 gallons per minute. The maximum surface temperatures were limited to 200°F to avoid water boiling at the surface. The values of the convective heat transfer coefficient were calculated with and without boundary layer control, and the graphs were drawn between 'h' and 'W' on logarithmic coordinates for different test pieces. Such boundary layer control gives a higher value of convective heat transfer from 25% to 140% compared with the water flow on both the outside and inside of the pipe surface simultaneously, depending upon the rate of flow and the number of holes. The maximum increase in the heat transfer coefficient was obtained by 480 surface holes. This percentage increase is greater at low rates of flow.

# VII. APPENDICES

Appendix I




Calibration Curve for Honeywell Flowmeter

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# VITA

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